

AN INVESTIGATION OF SOME OF THE  
CHARACTERISTICS OF A JERK PUMP  
INJECTION SYSTEM FOR  
DIESEL ENGINES  
—•••••  
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An Investigation of Some of the Characteristics  
of a Jerk Pump Injection System for Diesel Engines

By

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B.S. (United States Naval Academy) 1930

Thesis

Submitted in partial satisfaction of the requirements for the degree of

MASTER OF SCIENCE

in

Mechanical Engineering

in the

Graduate Division

of the

University of California

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## STATEMENT OF PROBLEM

The purpose of this investigation was to determine the influence of pump speed, nozzle opening pressure, rack setting, and length of line on the discharge rate of the injection valve in a jerk pump injection system for Diesel engines. Since some of the above variables may be controlled by an operator for any given system, it is felt that any correlation, mathematical or experimental, between these several variables would be of value to the designer of the injection system as well as to the Diesel operator.

In addition to obtaining a correlation between the above variables, FOR A PARTICULAR INSTALLATION, it appeared desirable to determine whether or not these variables might be changed with respect to one another to obtain performance which might be predicted, and which would result in better operation.

With these objectives in mind the problem was attacked from a purely experimental point of view and the results are recorded herein.

EXPERIMENTAL INVESTIGATION

The purpose of this investigation was to determine the influence of pump speed, water quality, and length of line on the efficiency of the injection system. In a first series of tests (see Table I) the effect of pump speed on the efficiency of the injection system was determined. The results of these tests are shown in Table II. It was found that the efficiency of the injection system increased with increasing pump speed. The efficiency of the injection system was also found to be influenced by the length of the line. The results of these tests are shown in Table III. It was found that the efficiency of the injection system decreased with increasing length of the line. The results of these tests are shown in Table IV. It was found that the efficiency of the injection system decreased with increasing length of the line.

In addition to determining the efficiency of the injection system, the effect of water quality on the efficiency of the injection system was also determined. The results of these tests are shown in Table V. It was found that the efficiency of the injection system decreased with increasing water quality. The results of these tests are shown in Table VI. It was found that the efficiency of the injection system decreased with increasing water quality.

With these results in mind the problem was extended to a study of the effect of the length of the line on the efficiency of the injection system. The results of these tests are shown in Table VII. It was found that the efficiency of the injection system decreased with increasing length of the line.



## APPARATUS AND TEST PROCEDURE

A general view of the apparatus is shown in Figure 1, and a schematic diagram of the same apparatus is shown in Figure 4. In the above two figures the same letters are used to designate the same parts. Referring to either Figure 1 or Figure 4:

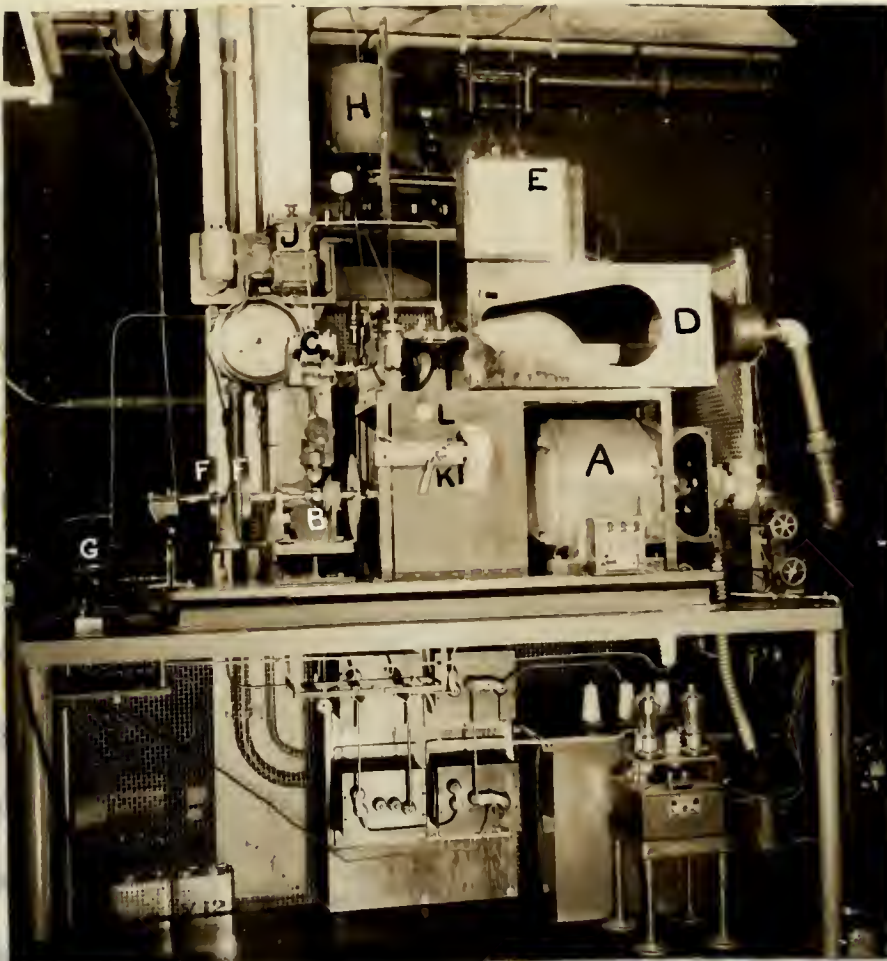


FIGURE 1.

"A" is a shunt motor having rheostatic speed control which drives the shaft to which is secured the cam "B", this cam in turn operates the plunger of the jerk pump "C". From the pump "C" the oil is discharged through a pipe line to the nozzle valve which is located in the box "D". A graduated scale is secured in box "D" as shown, by means of which the spray penetration may be determined. Mounted on

A general view of the structure is shown in Figure 1, and a  
 detailed diagram of the same apparatus is shown in Figure 2. In  
 the above two figures the main features are used to designate the  
 same parts. Indicated in Figure 1 are Figure 2.



FIGURE 1.

"A" is a short metal having resistance equal to that of the  
 the shell to which is secured the end "B", this end is then secured  
 the bottom of the pipe joint "C". From the joint "C" the oil is dis-  
 charged through a pipe line to the inside valve which is located in  
 the joint "D". A graduated scale is secured to the "D" as shown by  
 means of which the spring mechanism may be determined. Indicated on



top of the box "D" is a stroboscopic neon light "E" which is made to illuminate the interior of the box "D" through a glass window in the top of "D". The time of the flashing of the neon light "E" may be made to occur at any desired angular shaft position by means of the graduated and variable rotary spark gap "F", one part of which is secured to the end of the motor shaft, while the variable part is mounted on the frame. A wiring diagram, Figure 5, shows the electrical connections between the spark gap "F" and the neon light "E". "H" is the oil reservoir located on one platform of a balance scale. Balance is obtained by placing weights of various magnitudes on the other platform; the instant of perfect balance being indicated by the flashing of a small neon light "L", Figure 1. This neon light is controlled by an electrical circuit through two small wires secured to either platform; the wires moving in and out of mercury baths, located under the platforms, as the balance changes position. From "H" the oil is delivered to the fuel pump "C" through the oil filter "J". It is to be noted that oil may be delivered to the pump "C" by gravity or under pressure by means of the variable speed pump "G". A pressure of 12 pounds per square inch was maintained at the suction side of the pump "C" throughout the investigation.

The pump used was a 10-millimeter "jerk-pump", having the conventional plunger scroll control. A cut-away picture of this type of pump is shown in Figure 2(B). The quantity of oil delivered by the pump may be varied by rotating the plunger "P" by means of the gear "G" which is secured to the plunger "P". Gear "G" is in turn

top of the box "1" is a cylindrical cover light "2" which is made  
 to illuminate the interior of the box "1" through a glass window  
 in the top of "1". The base of the cylinder of the cover light "2"  
 will be made to cover all the desired lighting effect resulting in making  
 of the illuminated and unilluminated areas only "2", the part of  
 which is covered by the top of the cover light, while the unilluminated  
 part is covered by the base. A window "3" is made in the side of  
 the electrical connections between the cover light "2" and the box  
 light "1". "2" is the oil reservoir located at the bottom of a  
 balance scale. Balance is obtained by placing weights on various  
 weights on the other platform. The constant of weight balance  
 being indicated by the flashing of a small lamp light "4", Figure 1.  
 This lamp light is controlled by an electrical circuit through two  
 small wires secured to either platform. The wires extend to and  
 are at various points, located under the platform, at the bottom  
 of the platform. From "5" the oil is delivered to the fuel pump  
 "6" through an oil filter "7". It is to be noted that all oil is  
 delivered to the pump "6" by gravity or under pressure by means of  
 the electric pump "8". A pressure of 15 pounds per square  
 inch was maintained at the various sides of the pump "6" throughout  
 the investigation.

The fuel used was a 10-40-100 "70-70-70", being the same  
 standard lightest weight control. A one-way pressure of 100 lbs  
 of fuel is shown in Figure 1(a). The quantity of oil delivered by  
 the pump was for testing in testing the pump "6" by means of the  
 pump "6" which is secured to the platform "7". Gas "9" is in test



actuated by means of an engaging rack, the position of which is controlled by the micrometer head "R", Figure 4. The end of the plunger "P" is actuated by the cam "B", Figure 4, by means of suitable linkage. The operation of the fuel pump "B", Figure 2, is as follows: as the plunger "P" moves from right to left, the space "E" becomes isolated when the flat plunger face reaches the left edge of the inlet channel "J". Further motion of "P" to the left causes the plunger to force oil from "E" past the check valve "V" until the left edge of the scroll space "S" reaches the right edge of the inlet channel "J". At this point the space "E" communicates directly with the inlet channel "J" through a small hole drilled from the plunger face to the scroll space "S", and the oil in space "E" is by-passed back to "J".

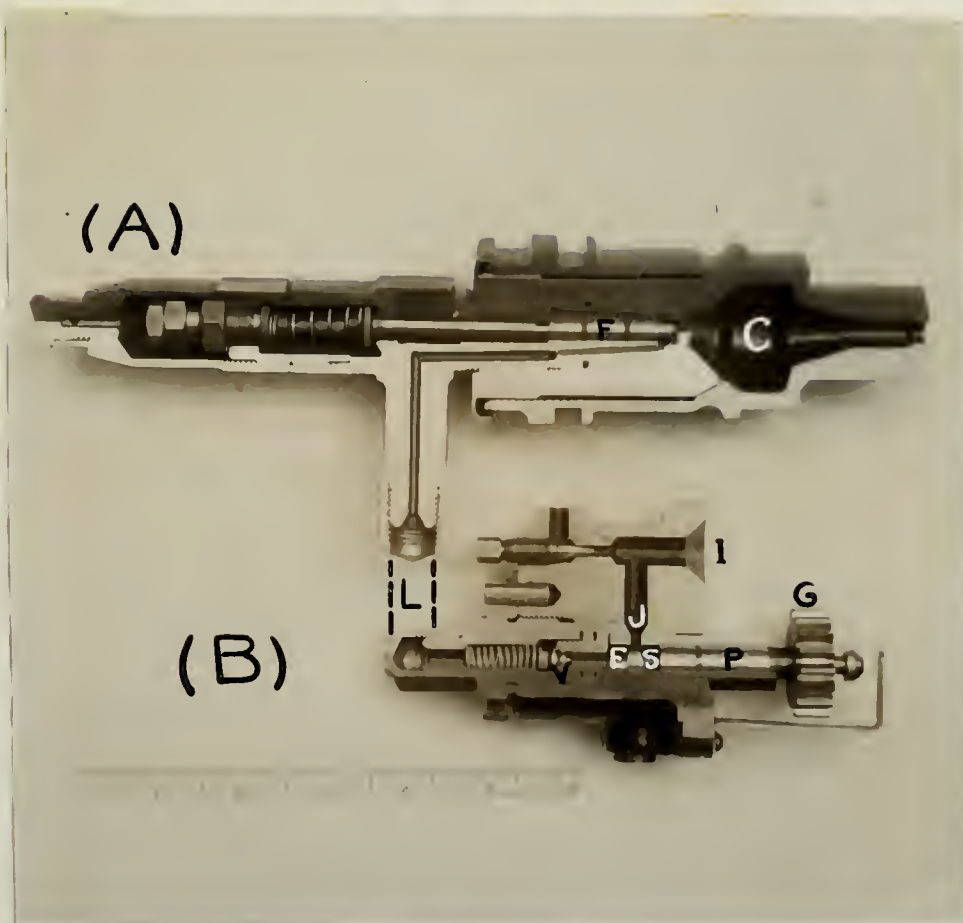


Figure 2.

entrance to each of the two pipes, the position of which is also  
 marked by the dimension lines  $2''$ , figure 4. The end of the pipes  
 $1''$  is indicated by the line  $1''$ , figure 4, by means of vertical lines.  
 The operation of the two pipes  $1''$ , figure 4, is as follows: as the  
 plunger  $1''$  moves from right to left, the space  $1''$  between the  
 ends of the two pipes (the space between the left edge of the left channel  
 $1''$ , figure 4, and the left edge of the right channel  $1''$ , figure 4) is  
 all then  $1''$  and the space  $1''$  still the left edge of the  
 right channel  $1''$  reaches the right edge of the left channel  $1''$ . In  
 this position the space  $1''$  communicates directly with the left channel  
 $1''$  through a small hole drilled from the plunger into the small  
 space  $1''$ , and the air in space  $1''$  is by-passed from  $1''$ .





Thus in this type of pump, delivery will always commence at the same angular shaft position, regardless of rack setting, whereas the angular shaft position at which oil ceases to be delivered will depend upon the setting of the rack.

Oil enters the pump Figure 2(B) at "I", is discharged past the check valve "V" from whence it is delivered to the nozzle valve, Figure 2(A) through the pipe line "L".

A Caterpillar Fuel Injection Valve designed for engines of  $5\frac{1}{4}$ " and  $5\text{-}3\frac{3}{4}$ " cylinder bore was used in this investigation. A picture of this valve, in cut-away section, is shown in Figure 2(A); the valve actually used, however, differs from the one shown in Figure 2(A) in that the pre-combustion chamber "C" was not used. The manufacturer's part designation for this valve is as listed below:

<u>Part Name</u>	<u>Part Number</u>	<u>Weight (Grams)</u>
Spray valve spring	1A6926	11.22
Spray valve spring stem	2A4684	16.07
Spray valve needle	2A4682	5.74

Other pertinent data pertaining to the nozzle valve are:

- (1) needle valve lift 0.007"; (2) needle valve stem diameter 0.039";
- (3) included angle between faces of valve seat =  $60^{\circ}$ ; (4) orifice length 0.118"; (5) orifice diameter 0.025"; (6) spring constant 771.2 lbs. per in. deflection; (7) clearances between "P" and "F" and their respective working surfaces = lap fit.

In obtaining the data necessary for plotting the curves shown

in Figure 1, 2 and 3, the adjustable spring of the injection valve  
was fixed at 100 lb. and the air in a small pump to cause the  
valve to open at the desired pressure. The pump used was equivalent  
of a constant volume and the number of pump strokes required  
for the delivery of a known weight of oil, as indicated by the balance  
and indicator light system, was recorded on the provided number "1".  
Figures 1 and 2. Several "runs" were made for each setting in  
order to obtain a series of data.

The spray operation shown in Figure 3 was taken at  
a pump speed of 800 R.P.M. and a tank setting of 100 (indicator load).  
These in Figure 3 were taken at the same pump speed but for a tank  
setting of 200 (light load). Inside opening pressure in both cases  
was 1700 lbs. per sq. in. Figures were taken for every one degree  
regular about displacement, starting with the first stroke of the  
injection and continuing until the point of cut-off. Since a speed  
of 800 R.P.M. is equivalent to one revolution in a tank of a second,  
the pictures were given a time exposure of one-half of a second.  
Spray giving only one injection per picture rather than a series  
of injections. The series of pictures represent several hundred  
injections rather than a time development of one injection.

The characteristic properties of the diesel fuel used were as

Following:  
Gravity..... 30.8 A.P.I. at 60°F.  
Viscosity..... 28.7 A.P.V. at 100°F.  
Sulfur Content..... 22.8 grains per 100 lb.



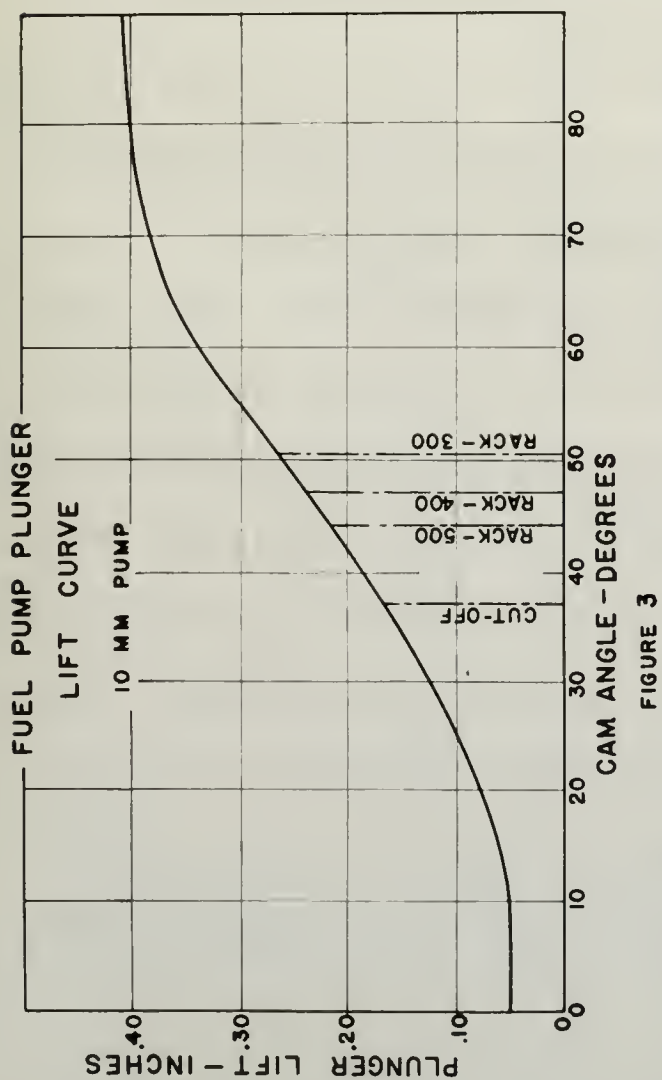


FIGURE 3

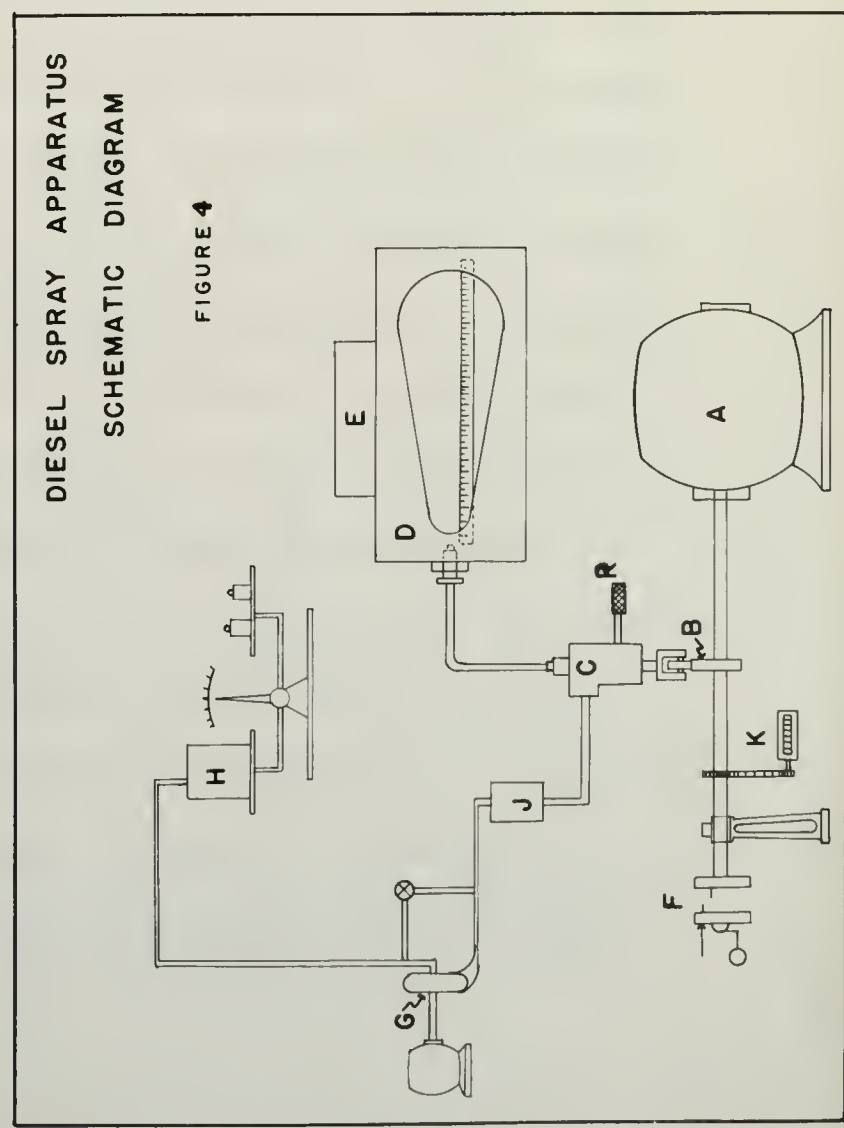


FIGURE 4

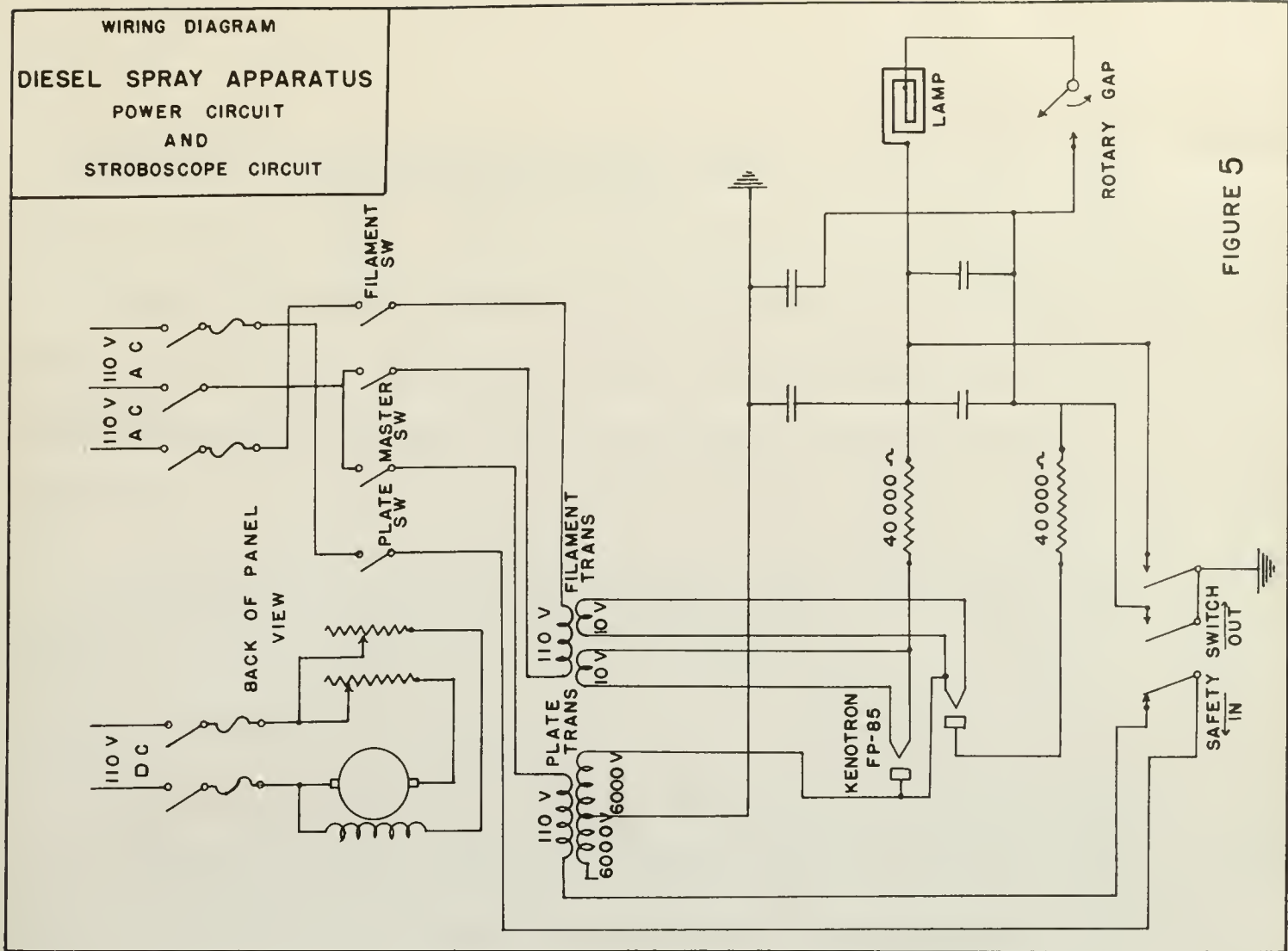


FIGURE 5





## DISCUSSION AND RESULTS

The quantity of oil which the fuel pump, Figure 2(B), should deliver for various rack settings, based on displacement data, was computed as follows and is shown by the plane surfaces A B C D in Figure 6:

Dial indicator readings were carefully taken of the cam contour for the various angular crank positions from which the lift curve, Figure 3, was constructed. Then the angular shaft position for cut-off (the point at which the fuel pump, Figure 2(B), began delivery) was obtained by manually unseating the check valve and observing the point at which the oil ceased to flow from the fuel pump; the oil to the suction side of the pump being maintained under pressure due to the gravity head from supply tank "H", see Figure 1. This angular shaft position was, as should be expected, the same for all rack settings. Next, the angular position of the shaft at the point of release (that point at which the pump stopped delivery due to the scroll position), was determined for each rack position. This was accomplished by again lifting the fuel pump check valve manually, and observing the angular shaft position at which oil began to flow from the pump. These angular positions were then transferred to the lift curve, Figure 3, from which the effective pump stroke for any rack setting was obtained by subtracting the lift at the point of cut-off from the lift at the point of release. Knowing the effective pump stroke, for any rack setting, the pump plunger area and the density

DISCUSSION AND CONCLUSIONS

The quantity of oil which the pump (Figure 1) would deliver for various crank positions, based on displacement data, was computed as follows and is shown in the table between A & B on Figure 2:

Dial indicator readings were carefully taken of the cam surface for the various angular crank positions from which the lift curve, Figure 2, was constructed. From the angular shaft position the lift at the point at which the fuel pump (Figure 1) began delivery was obtained by mentally measuring the shaft angle and observing the point at which the oil ceased to flow from the fuel pump. The lift for the suction side of the pump being contained under pressure due to the gravity head from supply tank "B", see Figure 1. This angular shaft position was, as should be expected, the same for all lift curves. Next, the angular position of the shaft at the point of release (that point at which the pump stopped delivery due to the supply position), was determined for each crank position. This was accomplished by again lifting the fuel pump check valve manually, and observing the angular shaft position at which oil began to flow from the pump. These angular positions were then transferred to the lift curve, Figure 2, from which the effective pump stroke for any crank setting was obtained by subtracting the lift at the point of cut-off from the lift at the point of release. Knowing the effective pump stroke, for any crank setting, the pump chamber area and the density



of the Diesel oil, the weight of oil discharged per stroke was computed from the formula: weight per stroke = area of piston x stroke of piston x oil density. Since the area of the pump was .000846 square feet (plunger diameter 10 millimeters), and the density of the oil at 12 lbs. per sq. in. pressure and 74°F, was 62.284 x .8676 = 54.038 pounds per cubic foot, the weight of oil delivered per stroke was: .000846 x 54.038 x  $\frac{\text{lift}''}{12}$  = .00381 x lift'' (lbs.). The results of the above computations are as tabulated:

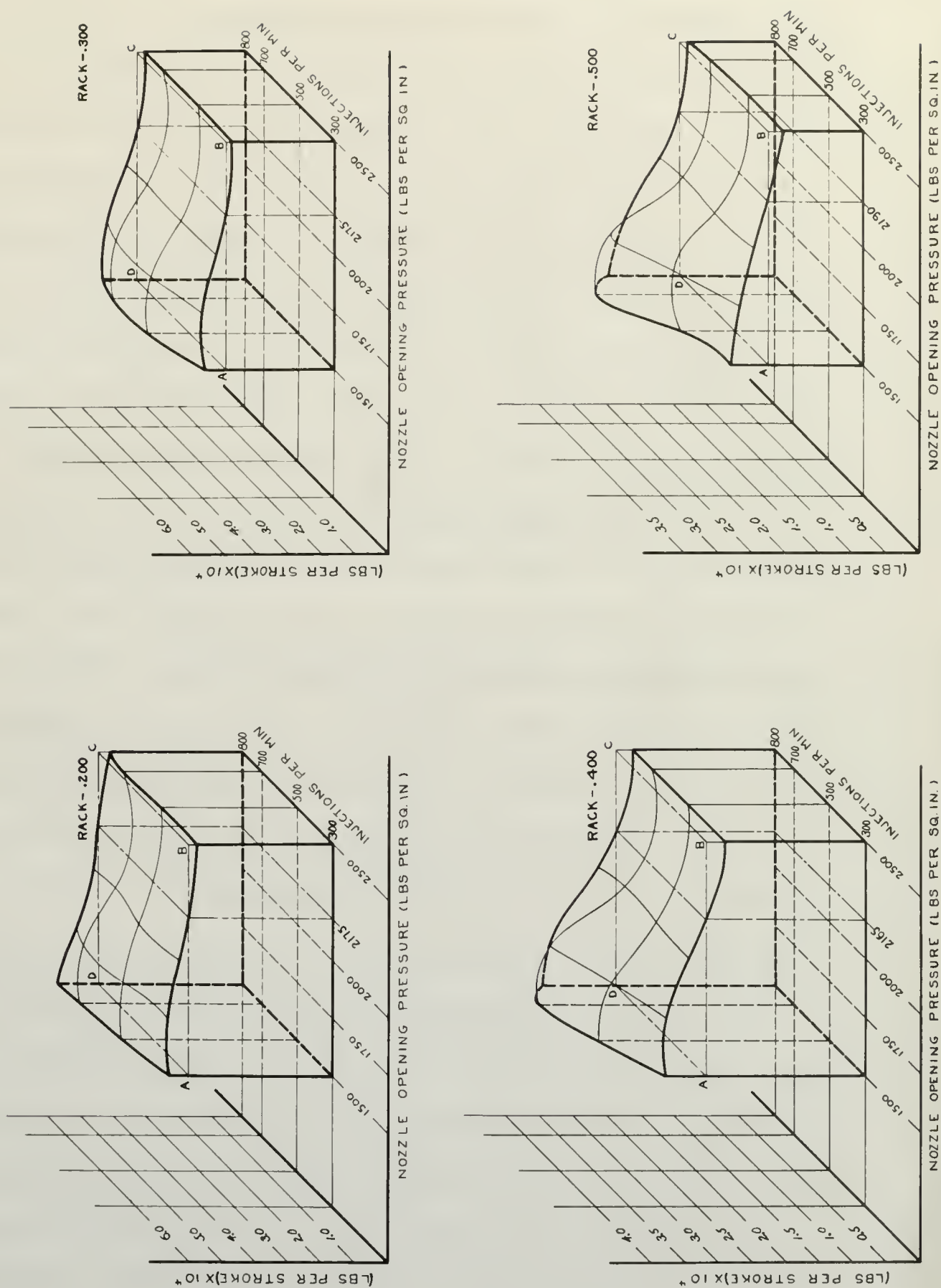
TABLE I

<u>Rack Setting</u>	<u>Cut Off</u>	<u>Re-lease</u>	<u>Lift at Cut Off</u>	<u>Lift at Release</u>	<u>Effective Pump Stroke</u>	<u>Calculated Wt. of oil per Stroke (lbs.)</u>
.150	376	56°	.175"	.305"	.130"	4.96 x 10 <sup>-4</sup>
.175	"	55°	"	.297"	.122"	4.65 x 10 <sup>-4</sup>
.200	"	54°	"	.290"	.115"	4.38 x 10 <sup>-4</sup>
.225	"	53°	"	.282"	.107"	4.08 x 10 <sup>-4</sup>
.250	"	52°	"	.274"	.099"	3.77 x 10 <sup>-4</sup>
.300	"	50.5°	"	.262"	.087"	3.315 x 10 <sup>-4</sup>
.350	"	49.25°	"	.253"	.078"	2.97 x 10 <sup>-4</sup>
.400	"	47.25°	"	.238"	.063"	2.40 x 10 <sup>-4</sup>
.450	"	46°	"	.227"	.052"	1.98 x 10 <sup>-4</sup>
.500	"	44°	"	.213"	.038"	1.45 x 10 <sup>-4</sup>
.550	"	42.5°	"	.202"	.027"	1.03 x 10 <sup>-4</sup>
.600	"	40°	"	.185"	.010"	0.381 x 10 <sup>-4</sup>

From Figure 6, it may be observed that the surface representing the oil actually discharged per stroke for any given rack setting



THE INFLUENCE OF PUMP SPEED AND NOZZLE  
OPENING PRESSURE  
ON DISCHARGE RATE FOR VARIOUS RACK SETTINGS  
PIPE LENGTH - 30 INCHES  
FIGURE 6







changes contour with both speed and pressure variations. Disregarding the dynamics of the system this surface should be coincident with the plane surface ABCD which represents the amount of oil which the pump should deliver based on displacement data. Obviously the planes A B C D in Figure 6 are horizontal as shown since the theoretical weight delivered per stroke is independent of all variables save rack setting. That the weight per stroke contour surface does not coincide with the plane surfaces A B C D Figure 6, is clearly shown for the rack settings considered (.200, .300, .400, .500). The fact that the pump actually delivers more (or less) oil than is shown by displacement computations may be explained in the following way.<sup>(1) (2) (6)\*</sup>

Once the check valve has been lifted off its seat, due to the oil pressure created by the plunger motion, oil will flow into the line and pressure waves will develop between the face of the pump plunger and the nozzle valve. When the pump reaches the point of release, the check valve will remain off its seat for an appreciable interval due to its inertia as well as to the friction force occasioned by the viscous drag of the oil flowing past the valve. Thus the next pressure wave, after release, reflected from the plunger face will force additional oil past the open check valve and into the fuel line. This action is possible, in spite of the fact that the plunger by-pass channel puts the pump chamber in direct communication with the suction side of the pump, since the by-pass channel area is so small that the pressure wave moving toward the plunger face will be reflected from the plunger face before it has driven any oil out of the chamber space into the suction line. After the check valve has been seated, pressure

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\*Such designations refer to similar numbers in bibliography.

valves will start before the end of the pump stroke and the check valve will not be of sufficient weight to keep the check valve against the oil pressure in the line, thus forcing oil from the pump barrel into the oil line. The higher the oil pressure in the line, the more the check valve opening pressure, the less will be the tendency for these pressure waves to re-open the check valve.

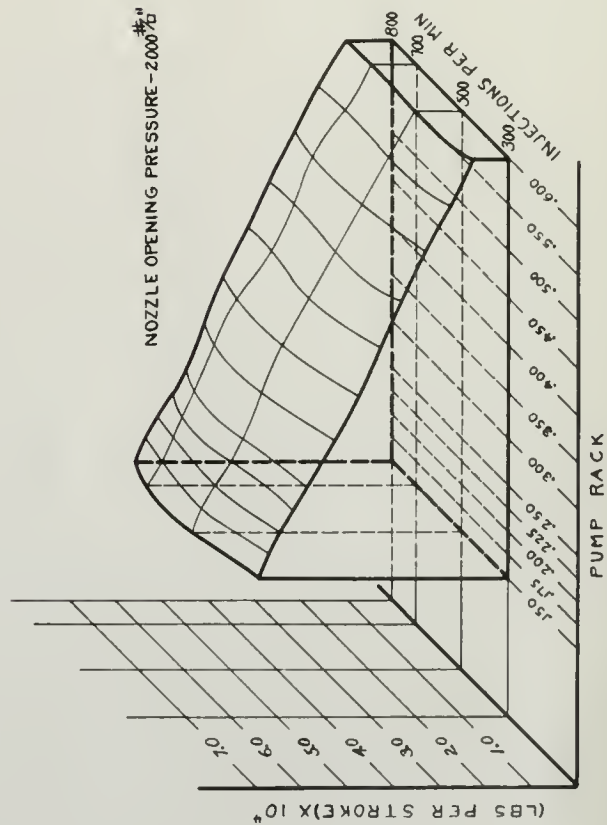
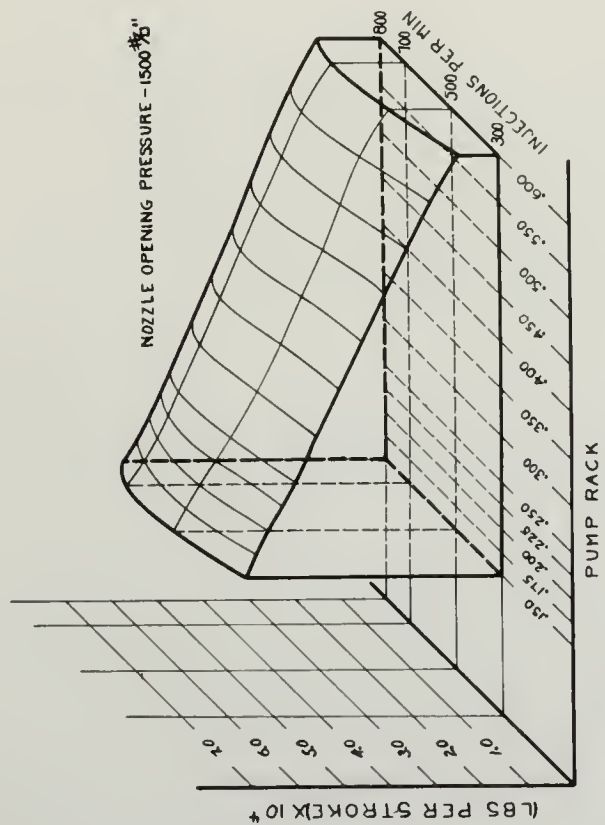
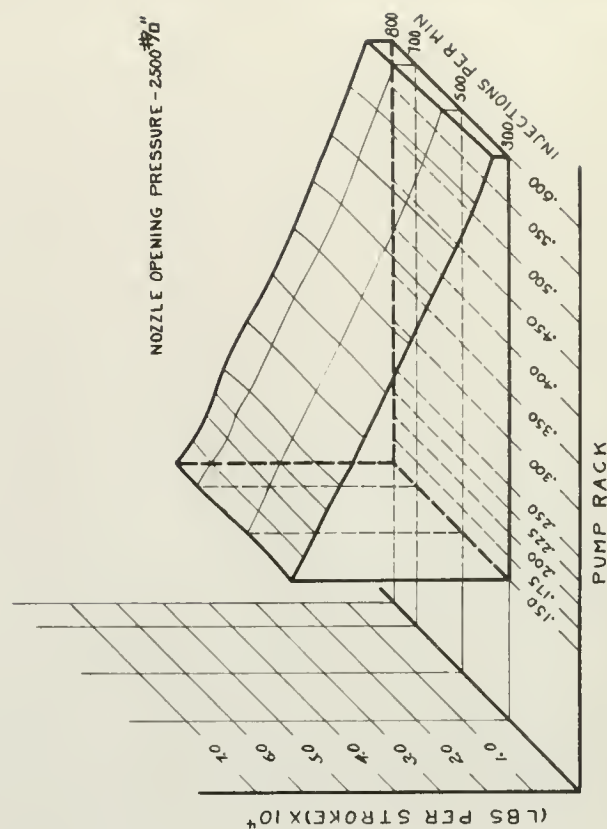
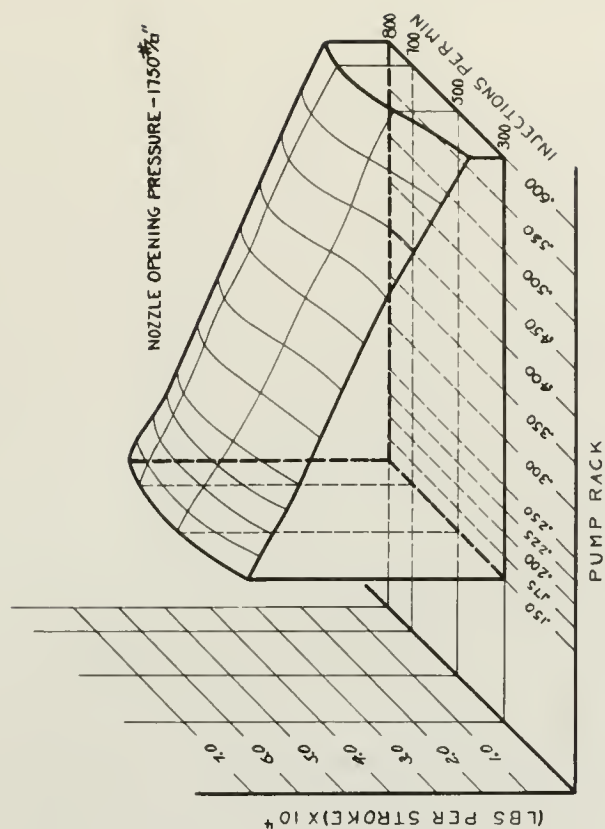
When it has been noted that the "super-charging" effect is already seen in Figure 5 for all test conditions and nozzle opening pressures below a certain value. The nozzle opening pressure at which "super-charging" does not exist are represented by the intersection of the lines between A & B, Figure 5, with their respective constant and zero, and these values are:

Back setting	Nozzle opening pressure (lb. per sq. in.)
200	117
250	117
300	117
350	117

It may be noted in Figure 5 that for the above nozzle opening pressures the actual quantity of oil delivered per stroke is independent of speed and varies linearly with back setting. As the nozzle opening pressure was increased beyond the average value of 218 lb. per sq. in., the oil pressure in the line was high enough, after the point of release, to keep the pump check valve closed against the action of the pressure surge in the pump barrel. Under these conditions, the pump discharged less oil than is indicated by the plate.

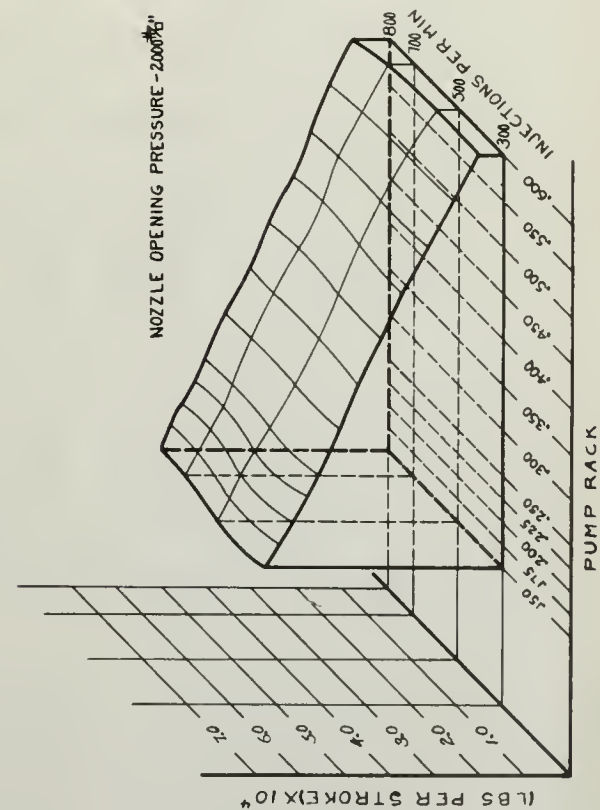
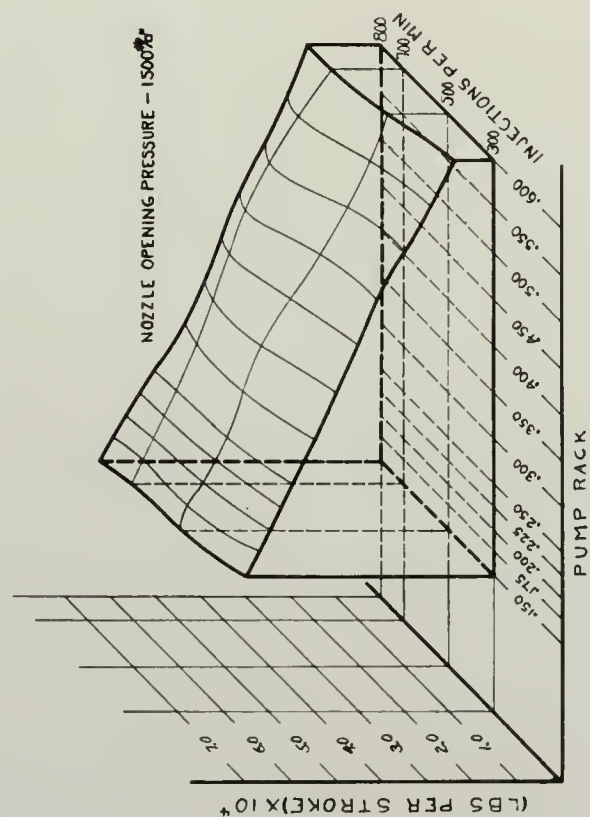
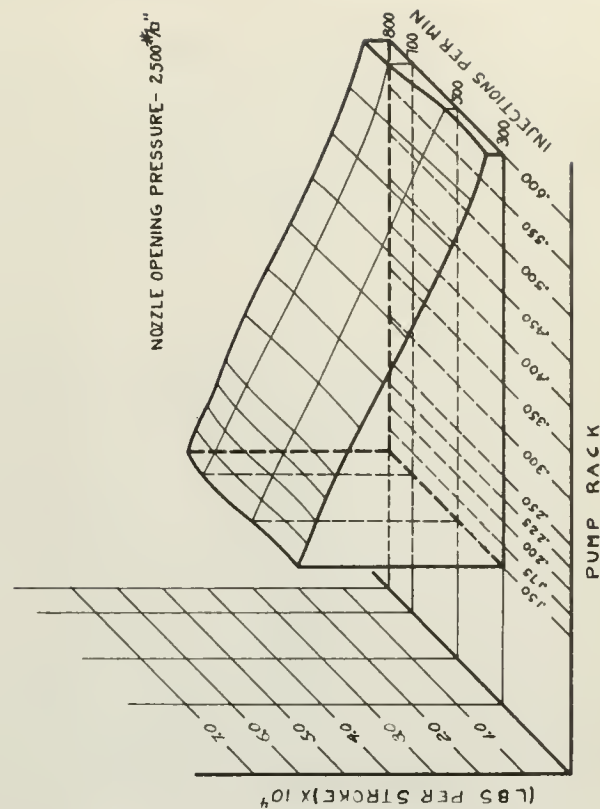
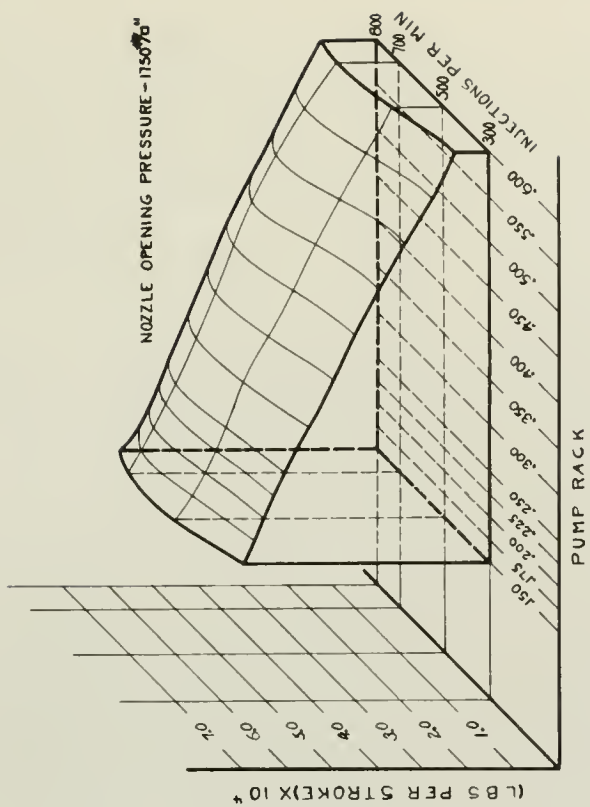


THE INFLUENCE OF PUMP SPEED AND RACK SETTING  
ON DISCHARGE RATE  
FOR VARIOUS NOZZLE OPENING PRESSURES  
PIPE LENGTH - 15 INCHES  
FIGURE 7





THE INFLUENCE OF PUMP SPEED AND RACK SETTING  
ON DISCHARGE RATE  
FOR VARIOUS NOZZLE OPENING PRESSURES  
PIPE LENGTH - 30 INCHES  
FIGURE 8





surfaces A B C D. The amount of oil actually delivered decreased with increasing nozzle opening pressures due to the greater force exerted by the oil to keep the check valve on its seat against the action of the pressure surges, and to the increased pump leakage occasioned by the higher discharge pressures.<sup>(5)</sup>

Figure 7 shows the influence of rack setting and speed on the weight of oil discharged per stroke for various nozzle opening pressures and a pipe length of 15 inches; Figure 8 shows a series of similar surfaces for a pipe length of 30 inches. Again it may be noted from these figures that as the nozzle opening pressure approaches 2175 lbs. per sq. in., the weight contour surface approaches plane surfaces; see Figures 7 and 8 for nozzle opening pressures of 2000 and 2500 lbs. per sq. in. These contour surfaces also clearly show the influence of pump speed changes on the quantity of oil discharged. Here it is to be noted that in general the weight of oil discharged increased with higher speed up to about 700 R.P.M.<sup>(3)</sup> while for the range between 700 -- 800 R.P.M., the weight of oil decreased;<sup>(5)</sup> this may be seen most clearly in Figures 7 and 8 and Table II for nozzle opening pressures of 1500 and 1750 lbs. per sq. in. For the nozzle opening pressures of 2000 and 2500 lbs. per sq. in. (in the neighborhood of 2175 lbs. per sq. in.) the influence of pump speed was not so marked.

Table II follows.



distance is 0.01. The amount of air actually flowing through  
 with increasing mass weight pressure due to the density force  
 caused by the air in the tank being in the tank against the  
 action of the pressure weight, and to the internal force weight  
 caused by the slight change pressure. (1)

Figure 7 shows the influence of mass weight and speed on the  
 weight of air flowing per stroke for various mass weight opera-  
 tions and a type length of 10 inches. Figure 8 shows a series of  
 similar curves for a type length of 20 inches. Again it may be  
 noted from these figures that as the mass weight increases ap-  
 proximately 15% the air weight increases 10% for 10 inches and  
 15% for 20 inches; and Figure 7 and 8 for mass weight increases of  
 1000 and 2000 lbs. per sq. in. These curves are also closely  
 show the influence of mass weight changes on the volume of air dis-  
 charged. Here it is to be noted that in general the weight of air  
 discharged increases with higher speed up to about 700 ft. (2)

Table II for the range between 700 -- 1000 ft. the weight of air dis-  
 charged, (3) this may be seen more clearly in Figures 7 and 8 and  
 Table II for mass weight increases of 1000 and 2000 lbs. per sq. in.  
 for the mass weight increases of 1000 and 2000 lbs. per sq. in.  
 (in the neighborhood of 15% the air weight) the influence of mass  
 weight on the air weight.

Table II follows.

TABLE II

Rack Setting	Pump Speed	ACTUAL DISCHARGE (LBS. PER STROKE X 10 <sup>0</sup> ); Pipe Length 30".			
		Nozzle Opening	Pressure (Lbs. Per	Sq. In.)	
		1500	1750	2000	2500
.200	300	498	505	467	418
	500	541	535	485	412
	700	556	550	450	424
	800	568	508	457	403
.300	300	394	404	366	318
	500	459	452	365	311
	700	460	457	359	312
	800	435	422	361	319
.400	300	303	304	269	216
	500	352	337	265	216
	700	388	373	257	217
	800	355	336	270	217
.500	300	202	191	166	122
	500	226	236	173	123
	700	300	279	167	128
	800	254	228	174	132

The fact that the quantity of oil increased for increased pump speed up to 700 R.P.M., and then decreased for further speed increases may be explained as follows: higher pump speeds give higher plunger speed, and hence impart greater velocities to the oil being discharged from the pump; this increase in kinetic energy causes a greater quantity of oil to flow past the pump check valve after release and before the check valve has become seated. As the pump speed increased beyond a certain value, however, the volumetric efficiency of the pump decreased since oil could not flow into the pump chamber fast enough to completely fill it, and the quantity of oil discharged decreased.

A comparison of Figures 7 and 8 shows pipe length, for the two

TABLE II

TOTAL DISCHARGE (GAL. PER MIN.) TO 14.7 PSIA					Flow Rate GPM	Total Discharge GPM
14.7 PSIA	15.0 PSIA	15.5 PSIA	16.0 PSIA	16.5 PSIA		
100	100	100	100	100	100	100
110	110	110	110	110		
120	120	120	120	120		
130	130	130	130	130		
140	140	140	140	140	200	200
150	150	150	150	150		
160	160	160	160	160		
170	170	170	170	170		
180	180	180	180	180	300	300
190	190	190	190	190		
200	200	200	200	200		
210	210	210	210	210		
220	220	220	220	220	400	400
230	230	230	230	230		
240	240	240	240	240		
250	250	250	250	250		
260	260	260	260	260	500	500
270	270	270	270	270		
280	280	280	280	280		
290	290	290	290	290		
300	300	300	300	300	600	600
310	310	310	310	310		
320	320	320	320	320		
330	330	330	330	330		
340	340	340	340	340	700	700
350	350	350	350	350		
360	360	360	360	360		
370	370	370	370	370		
380	380	380	380	380	800	800
390	390	390	390	390		
400	400	400	400	400		
410	410	410	410	410		
420	420	420	420	420	900	900
430	430	430	430	430		
440	440	440	440	440		
450	450	450	450	450		

The flow rate and quantity of all discharges for the above pump  
 tested up to 14.7 psia, and then decreased for higher pump pressures  
 and are explained as follows: As the pump speed is increased, the  
 speed, and hence the pressure, increases to the full design  
 from the pump; this increase in static pressure causes a greater quantity  
 of oil to flow from the pump than with other valves and before the  
 check valve has become seated. As the pump speed is increased beyond a  
 certain value, however, the resistance offered by the pump increases  
 and all oil in the line is used in the pump chamber and enough to completely  
 fill it, and the quantity of oil discharged decreases.

A comparison of Figures 1 and 2 shows pipe length for the two





Figure 9

SPRAY DEVELOPMENT FOR AVERAGE OF SEVERAL HUNDRED INJECTIONS.

Time Between Frames, .000278 seconds

Nozzle Opening Pressure, 1750 lbs. per sq. in.

Pump Speed, 600 R.P.M.

Length of Pipe Line, 15 inches

Rack Setting, .400 (2.40 x 10<sup>4</sup> lbs. of oil per stroke)

Chamber Pressure, Atmospheric





Figure 10

SPRAY DEVELOPMENT FOR AVERAGE OF SEVERAL HUNDRED INJECTIONS.

Time Between Frames, .000278 seconds  
 Nozzle Opening Pressure, 1750 lbs. per sq. in.  
 Pump Speed, 600 R.P.M.  
 Length of Pipe Line, 15 inches  
 Rack Setting, .600 ( $.381 \times 10^4$  lbs. of oil per stroke)  
 Chamber Pressure, Atmospheric.





lengths investigated, to have little influence.<sup>(3)</sup> For nozzle opening pressures of 1750 and 2500 lbs. per sq. in., the two pipe lengths gave almost identical contour surfaces, while for nozzle opening pressures of 1500 and 2000 lbs. per sq. in. some irregularities between the contour surfaces may be noted. These irregularities, as may be seen, occurring for pump speeds of 500 R.P.M. and above.

A comparison of Figures 9 and 10 show the influence of rack setting on penetration at a pump speed of 600 R.P.M., and a nozzle opening pressure of 1750 lbs. per sq. in., when discharging against atmospheric pressure. Figure 9 is for a rack setting of .400, while Figure 10 is for a rack setting of .600. For both rack settings, the pictures marked 1 were taken at two degrees of pump shaft angular displacement after the point of injection was observed which is equivalent to one eighteen-hundredth of a second. Thereafter the pictures were taken in succession at one degree pump shaft intervals (one thirty-six-hundredth of a second). In Figure 9, evidence of secondary discharges may be seen in pictures 4, 5, 13, 14, 15 and 17.<sup>(4)</sup> Maximum penetration of 16.5 inches is shown in picture 12 and cut off in picture 13. The depth of penetration is quite uniform up to the point of cut off, picture 13. It is interesting to note at this point that Figure 9 shows a definite injection period over an interval of 13 degrees angular pump shaft displacement, whereas the table on page 12 for a rack setting of .400 shows the injection period to be only 10.25 degrees. This point again established the fact that the pump actually discharges more oil under certain conditions than is theoretically possible.



The first of these is the fact that the  
 pressure of 1700 and 1800 lbs. per sq. in. is  
 not a constant value, but varies with the  
 rate of flow. The pressure is higher at  
 the inlet than at the outlet, and the  
 difference between the two is about 100  
 lbs. per sq. in. This difference is due  
 to the friction between the fluid and  
 the walls of the pipe.

A comparison of Figure 1 and Figure 2  
 will show that the pressure is higher at  
 the inlet than at the outlet, and the  
 difference between the two is about 100  
 lbs. per sq. in. This difference is due  
 to the friction between the fluid and  
 the walls of the pipe. The pressure is  
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 difference between the two is about 100  
 lbs. per sq. in. This difference is due  
 to the friction between the fluid and  
 the walls of the pipe.

In Figure 10 it may be readily seen that the spray penetration is quite irregular. Secondary discharges may be noted in pictures 3, 4, 5, 6, 7, 9 and 10. Cut off has taken place in picture 7, and here the maximum penetration was about 10 inches. Again it may be noted in Figure 10 that injection took place over a pump shaft interval of 8 degrees whereas the table on page 12, for a rack setting of .600, shows this interval to be only 3 degrees.

In figure 11 it can be readily seen that the upper portion of the  
 is quite irregular. The lower portion is in fact a flat  
 1, 2, 3, 4, 5, 6, 7, 8, 9, 10, 11, 12, 13, 14, 15, 16, 17, 18, 19, 20, 21, 22, 23, 24, 25, 26, 27, 28, 29, 30, 31, 32, 33, 34, 35, 36, 37, 38, 39, 40, 41, 42, 43, 44, 45, 46, 47, 48, 49, 50, 51, 52, 53, 54, 55, 56, 57, 58, 59, 60, 61, 62, 63, 64, 65, 66, 67, 68, 69, 70, 71, 72, 73, 74, 75, 76, 77, 78, 79, 80, 81, 82, 83, 84, 85, 86, 87, 88, 89, 90, 91, 92, 93, 94, 95, 96, 97, 98, 99, 100, 101, 102, 103, 104, 105, 106, 107, 108, 109, 110, 111, 112, 113, 114, 115, 116, 117, 118, 119, 120, 121, 122, 123, 124, 125, 126, 127, 128, 129, 130, 131, 132, 133, 134, 135, 136, 137, 138, 139, 140, 141, 142, 143, 144, 145, 146, 147, 148, 149, 150, 151, 152, 153, 154, 155, 156, 157, 158, 159, 160, 161, 162, 163, 164, 165, 166, 167, 168, 169, 170, 171, 172, 173, 174, 175, 176, 177, 178, 179, 180, 181, 182, 183, 184, 185, 186, 187, 188, 189, 190, 191, 192, 193, 194, 195, 196, 197, 198, 199, 200, 201, 202, 203, 204, 205, 206, 207, 208, 209, 210, 211, 212, 213, 214, 215, 216, 217, 218, 219, 220, 221, 222, 223, 224, 225, 226, 227, 228, 229, 230, 231, 232, 233, 234, 235, 236, 237, 238, 239, 240, 241, 242, 243, 244, 245, 246, 247, 248, 249, 250, 251, 252, 253, 254, 255, 256, 257, 258, 259, 260, 261, 262, 263, 264, 265, 266, 267, 268, 269, 270, 271, 272, 273, 274, 275, 276, 277, 278, 279, 280, 281, 282, 283, 284, 285, 286, 287, 288, 289, 290, 291, 292, 293, 294, 295, 296, 297, 298, 299, 300, 301, 302, 303, 304, 305, 306, 307, 308, 309, 310, 311, 312, 313, 314, 315, 316, 317, 318, 319, 320, 321, 322, 323, 324, 325, 326, 327, 328, 329, 330, 331, 332, 333, 334, 335, 336, 337, 338, 339, 340, 341, 342, 343, 344, 345, 346, 347, 348, 349, 350, 351, 352, 353, 354, 355, 356, 357, 358, 359, 360, 361, 362, 363, 364, 365, 366, 367, 368, 369, 370, 371, 372, 373, 374, 375, 376, 377, 378, 379, 380, 381, 382, 383, 384, 385, 386, 387, 388, 389, 390, 391, 392, 393, 394, 395, 396, 397, 398, 399, 400, 401, 402, 403, 404, 405, 406, 407, 408, 409, 410, 411, 412, 413, 414, 415, 416, 417, 418, 419, 420, 421, 422, 423, 424, 425, 426, 427, 428, 429, 430, 431, 432, 433, 434, 435, 436, 437, 438, 439, 440, 441, 442, 443, 444, 445, 446, 447, 448, 449, 450, 451, 452, 453, 454, 455, 456, 457, 458, 459, 460, 461, 462, 463, 464, 465, 466, 467, 468, 469, 470, 471, 472, 473, 474, 475, 476, 477, 478, 479, 480, 481, 482, 483, 484, 485, 486, 487, 488, 489, 490, 491, 492, 493, 494, 495, 496, 497, 498, 499, 500, 501, 502, 503, 504, 505, 506, 507, 508, 509, 510, 511, 512, 513, 514, 515, 516, 517, 518, 519, 520, 521, 522, 523, 524, 525, 526, 527, 528, 529, 530, 531, 532, 533, 534, 535, 536, 537, 538, 539, 540, 541, 542, 543, 544, 545, 546, 547, 548, 549, 550, 551, 552, 553, 554, 555, 556, 557, 558, 559, 560, 561, 562, 563, 564, 565, 566, 567, 568, 569, 570, 571, 572, 573, 574, 575, 576, 577, 578, 579, 580, 581, 582, 583, 584, 585, 586, 587, 588, 589, 590, 591, 592, 593, 594, 595, 596, 597, 598, 599, 600, 601, 602, 603, 604, 605, 606, 607, 608, 609, 610, 611, 612, 613, 614, 615, 616, 617, 618, 619, 620, 621, 622, 623, 624, 625, 626, 627, 628, 629, 630, 631, 632, 633, 634, 635, 636, 637, 638, 639, 640, 641, 642, 643, 644, 645, 646, 647, 648, 649, 650, 651, 652, 653, 654, 655, 656, 657, 658, 659, 660, 661, 662, 663, 664, 665, 666, 667, 668, 669, 670, 671, 672, 673, 674, 675, 676, 677, 678, 679, 680, 681, 682, 683, 684, 685, 686, 687, 688, 689, 690, 691, 692, 693, 694, 695, 696, 697, 698, 699, 700, 701, 702, 703, 704, 705, 706, 707, 708, 709, 710, 711, 712, 713, 714, 715, 716, 717, 718, 719, 720, 721, 722, 723, 724, 725, 726, 727, 728, 729, 730, 731, 732, 733, 734, 735, 736, 737, 738, 739, 740, 741, 742, 743, 744, 745, 746, 747, 748, 749, 750, 751, 752, 753, 754, 755, 756, 757, 758, 759, 760, 761, 762, 763, 764, 765, 766, 767, 768, 769, 770, 771, 772, 773, 774, 775, 776, 777, 778, 779, 780, 781, 782, 783, 784, 785, 786, 787, 788, 789, 790, 791, 792, 793, 794, 795, 796, 797, 798, 799, 800, 801, 802, 803, 804, 805, 806, 807, 808, 809, 810, 811, 812, 813, 814, 815, 816, 817, 818, 819, 820, 821, 822, 823, 824, 825, 826, 827, 828, 829, 830, 831, 832, 833, 834, 835, 836, 837, 838, 839, 840, 841, 842, 843, 844, 845, 846, 847, 848, 849, 850, 851, 852, 853, 854, 855, 856, 857, 858, 859, 860, 861, 862, 863, 864, 865, 866, 867, 868, 869, 870, 871, 872, 873, 874, 875, 876, 877, 878, 879, 880, 881, 882, 883, 884, 885, 886, 887, 888, 889, 890, 891, 892, 893, 894, 895, 896, 897, 898, 899, 900, 901, 902, 903, 904, 905, 906, 907, 908, 909, 910, 911, 912, 913, 914, 915, 916, 917, 918, 919, 920, 921, 922, 923, 924, 925, 926, 927, 928, 929, 930, 931, 932, 933, 934, 935, 936, 937, 938, 939, 940, 941, 942, 943, 944, 945, 946, 947, 948, 949, 950, 951, 952, 953, 954, 955, 956, 957, 958, 959, 960, 961, 962, 963, 964, 965, 966, 967, 968, 969, 970, 971, 972, 973, 974, 975, 976, 977, 978, 979, 980, 981, 982, 983, 984, 985, 986, 987, 988, 989, 990, 991, 992, 993, 994, 995, 996, 997, 998, 999, 1000.



## CONCLUSIONS

1. From Figure 6 it may be concluded that for this system there exists a certain nozzle opening pressure for which a linear relation exists between weight of fuel discharged per stroke and rack setting, and further that this linear relation is independent of speed for the range investigated; that is to say, for a pump speed from 300 -- 800 R.P.M. and a nozzle opening pressure of from 1500 -- 2500 pounds per sq. in. In this case the nozzle opening pressure required to give this linear relation was found to be about 2175 pounds per sq. in.

It appears reasonable to suppose that a similar nozzle opening pressure will exist, and may be found, for any injection system of the type investigated.

2. A comparison of Figures 7 and 8 shows that for the pipe lengths investigated there was little or no change in the weight of oil delivered per stroke for the two pipe lengths.

3. Figures 6, 7 and 8 show definitely that the quantity of oil discharged per stroke decreases with increase in the nozzle opening pressure.

4. Figures 6, 7 and 8 show that when operating at a nozzle opening pressure other than 2175 pounds per sq. in., as mentioned in 1 above, the weight of oil discharged per stroke will vary with pump speed, and the greater the departure of the pressure



# DISCUSSION

1. From Figure 1 it can be seen that the data points  
show a definite trend towards increasing pressure for small values  
of the parameter  $\alpha$  and a slight decrease for large values of  $\alpha$ .  
This is in agreement with the theoretical prediction that the  
pressure should increase with  $\alpha$  for small values of  $\alpha$  and  
decrease for large values of  $\alpha$ . The data points for  $\alpha = 0.1$   
and  $\alpha = 0.2$  are shown in Figure 1. The data points for  
 $\alpha = 0.1$  are shown in Figure 2. The data points for  
 $\alpha = 0.2$  are shown in Figure 3. The data points for  
 $\alpha = 0.1$  and  $\alpha = 0.2$  are shown in Figure 4.

2. It is seen from Figure 1 that the data points  
show a definite trend towards increasing pressure for small  
values of  $\alpha$  and a slight decrease for large values of  $\alpha$ .  
This is in agreement with the theoretical prediction that the  
pressure should increase with  $\alpha$  for small values of  $\alpha$  and  
decrease for large values of  $\alpha$ .

3. A comparison of Figure 1 and 2 shows that for the  
same values of  $\alpha$  the data points for Figure 1 are higher  
than those for Figure 2. This is in agreement with the  
theoretical prediction that the pressure should increase with  
the value of  $\alpha$ .

4. Figure 5, 6 and 7 show that the data points  
for Figure 5 are higher than those for Figure 6 and 7.  
This is in agreement with the theoretical prediction that the  
pressure should increase with the value of  $\alpha$ .

5. Figure 8, 9 and 10 show that the data points  
for Figure 8 are higher than those for Figure 9 and 10.  
This is in agreement with the theoretical prediction that the  
pressure should increase with the value of  $\alpha$ . It is also  
seen from Figure 8, 9 and 10 that the data points for  
Figure 8 are higher than those for Figure 9 and 10.

from this value of nozzle opening pressure, the greater will become the influence of pump speed. Generally speaking, the quantity of oil delivered per stroke, for the system investigated, increased with higher pump speed up to about 700 R.P.M., and further speed increase caused the quantity of oil discharged to decrease.

5. The quantity of oil delivered per stroke follows in a general way the rack setting, as seen from Figures 7 and 8, but this relation is not linear unless the nozzle is set to open at the proper spring setting (2175 pounds per sq. in. for this system); see Figure 6.

6. For a moderate load (rack setting of .400), it may be noted from Figure 9 that the spray penetration is reasonably uniform and that there is little evidence of secondary discharges. However, for light loads (rack setting of .600), it may be seen from Figure 10 that the spray penetration is irregular and that there is much evidence of secondary discharges.

from this series of results appears to be that the quantity of the influence of each factor, especially frequency, the quantity of all factors for which, for the purpose of statistical treatment, this factor may be taken up as a unit, the factor which has been used the quantity of all factors for which.

2. The quantity of all factors for which follows in a general way the same pattern, as seen from figures 1 and 2, but this relation is not linear, which has been seen in the proper order of the factors (the factors are not in the same order as in figure 1).

3. For a moderate load (load rating of 100), it may be seen from figure 3 that the frequency of vibration is reasonably constant and that there is little evidence of secondary vibration. However, for light loads (load rating of 100), it may be seen from figure 4 that the frequency of vibration is irregular and that there is some evidence of secondary vibration.

4. For a heavy load (load rating of 200), it may be seen from figure 5 that the frequency of vibration is irregular and that there is some evidence of secondary vibration. However, for very heavy loads (load rating of 300), it may be seen from figure 6 that the frequency of vibration is irregular and that there is some evidence of secondary vibration.



### ACKNOWLEDGMENT

The author desires especially to thank Professor C. J. Vogt of the Department of Mechanical Engineering, University of California, who started this investigation and who has given untiringly and cheerfully of both his time and wisdom in order that this problem might be carried forward.

Thanks are also extended to Lieutenant H. S. Persons, U.S.N., who assisted in obtaining all the necessary data and the taking and developing of the pictures contained herein; and to Mr. J. E. Gullberg, Lecturer in the Department of Zoology, University of California, for the assistance given in the taking of many pictures.

Finally, I desire to thank members of the W. P. A. Project in the Department of Mechanical Engineering for their work in constructing the experimental station and making necessary alterations.



APPENDIX

The author has been especially fortunate in the cooperation of the Department of Industrial Engineering, University of California, Berkeley, who provided the facilities and the personnel for the study of the problem and observation of both the line and shop in which this problem was being solved.

Thanks are also due to the Department of Industrial Engineering, University of California, Berkeley, who assisted in obtaining all the necessary data and the facilities and equipment of the various machines involved and to Dr. H. B. Gilbreth, Professor in the Department of Biology, University of California, for the assistance given in the field of shop management.

Finally, I desire to thank the Department of Industrial Engineering, University of California, Berkeley, for the facilities and equipment provided for the study of the problem and observation of both the line and shop in which this problem was being solved.

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APPENDIX

- (1) The following information is from the report of the "Joint Committee on the Administration of Justice" of 1911.
- (2) The following information is from the report of the "Joint Committee on the Administration of Justice" of 1911.
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